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A non-conventional interpretation of thermal regeneration in steam cycles

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ABSTRACT

The paper aims to contribute to a better understanding of the thermodynamic concept of heat regeneration in steam power plants with a finite number of bleedings. A regenerative Rankine cycle is compared to a complex system (CHC – complete hybrid cycle) composed by one non-regenerative Rankine cycle (HEC – hybrid engine cycle) and more reverse cycles (*RCs* – reverse cycles), as many as the number of the bleedings, able to pump heat from the condenser to a series of surface feedwater heaters, disposed upstream of the steam plant boiler. The COPs (coefficients of performance) of the heat pumps are evaluated, and new interesting formulations of the efficiency of the regenerative steam cycle are proposed.

In particular a steam cycle with two bleedings is analyzed, neglecting heat losses and pressure drops in the boiler and considering irreversibility only along the expansion line of the steam turbine and into the feedwater heaters. The efficiency and the work of the regenerative cycle are compared to the analogous values of the CHC cycle composed by one simple steam cycle (HEC) and two heat pump cycles (*RCs*), with steam as the working fluid. The two reverse cycles are considered completely reversible and raising heat from the condenser temperature to the bled steam condensing temperature. The paper shows the most significant results of the study in order to analyze the regenerative cycle and the CHC cycle in comparison with the non-regenerative Rankine cycle; in particular, the analysis is focused on the evaluation of the useful work, the heat supplied and the heat rejected for the examined cycles.

1. Introduction

The high efficiency values reached nowadays by steam power plants are due, mainly, to two factors: improvements to materials, allowing the increase of temperatures and pressures, and thermodynamic cycle improvements [1–6]. Among these last, it is important to cite: the condenser vacuum decrease, the increase of the steam cycle maximum temperature and pressure, regenerative feed-heating and reheating [1–3,6–8]; the application of all these techniques allows the construction of ultra-super-critical steam power plants characterized by very high global efficiencies, near 47%, and at present there are many outstanding projects aiming at still higher efficiency values [3,6].

Certainly, in this context, a primary role is played by thermal regeneration that, as known, consists in extracting, at different pressure levels, part of the working fluid flow rate during the expansion in turbine, and using it for heating the feedwater in a train of feedwater heaters [1,2,7–15]. As well known and widely explained in literature [1,2,8,9], the advantage provided to power plant efficiency by a finite number of bleedings can be evaluated considering the following two distinct concomitant and contrasting thermodynamic effects of the thermal regeneration: on one side, the reduction of the sources multiplicity factor, achieved raising the mean temperature of heat reception by water in the boiler; on the other side, the irreversibility introduced by the heat exchange in heaters, characterized by a finite temperature difference between feedwater and bled steam [1,2,7,12]. As a consequence, considering the same water temperature at the boiler inlet, it is convenient to increase the number of bleedings in order to reduce the irreversibility into the feedwater heaters [1,2,7,12]. Several studies, based on this observation, lead to the conclusion that increasing the number of steam bleedings from the turbine and optimizing their pressure produces a considerable increase in the cycle thermodynamic efficiency [16–19].

The limit case is that of a “continuous regeneration”, in which an infinite number of bleedings causes the feed-water temperature rise, analogous to the ideal case described by Haywood in [1], which consists of passing the feedwater coming from the condenser through an infinite number of coils placed between successive pairs of an infinite number of turbine stages. Both cases, obviously not realizable, are characterized by a null temperature difference between the exchanging fluids, thus the maximum efficiency increase coincides with the maximum sources multiplicity factor reduction; it is quite correct, of course, only if the steam is bled from the saturated zone of the turbine expansion.

In this paper, the authors have faced the issue of thermal regeneration from a non-conventional point of view, taking as reference

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Nomenclature

Symbol	Description (Units)
COP	coefficient of performance (–)
H, h	enthalpy (J/kg)
L	work (J)
m	mass (kg)
Q_1	heat supplied (J)
Q_2	heat rejected (J)
R	regeneration degree
Greek symbol	
η	efficiency (–)
φ	reciprocal of *NR* cycle efficiency (–)
Subscript	
0	condenser outlet
1	pump exit
2	economiser outlet
2′	evaporator outlet
3	superheater outlet
4_s, 4	steam turbine outlet
a_i	feedwater heaters outlet (a_x, a_y)
e_i	hydraulic turbine outlet (e_x, e_y)
HC	hybrid cycle
i	i^{th} bleeding
n	number of bleedings
NR	non-regenerative
R	regenerative
RC	reverse cycle
T	turbine
X, Y	bled steam
x, y	condensed bled steam

and further developing the study proposed by Bignardi and Trucco in [12] with the aim of proposing a particular intuitive method.

In particular, a regenerative steam power plant constituted by n steam bleedings from the turbine is analyzed in order to demonstrate its equivalence, from the energy balance point of view, to a "hybrid plant", composed by one conventional non-regenerative Rankine cycle and n heat pump cycles, carrying heat from the condenser to the feedwater into n surface heaters. To confirm the validity of the study, the paper describes the main results concerning the energy balances of non-regenerative, regenerative and hybrid steam cycles; in particular, the effect of different steam turbine isentropic efficiency values and the effect of the "coefficient of performance" of the heat pumps on the plants performance are highlighted.

2. The proposed criterion for the analysis of the thermal regeneration

The present paragraph describes the criterion proposed by the authors to explain the advantages of the thermal regeneration on the steam cycle efficiency. The proposed method has been developed in order to compare the results of this relatively more intuitive approach with the classical theories. To explain the method, some considerations have to be presented.

Supposing the same mass flowing in the boiler, equal to $(1+\Sigma_i m_i)$, and identical steam thermodynamic conditions at the boiler exit (state 3) and condenser inlet (state 4s), a regenerative (R) and a non-regenerative (NR) steam power plant are compared by the different energy balances point of view. For the regenerative cycle it is possible to observe a contemporaneous reduction of the heat supplied Q_1, the work L and the rejected heat Q_2:

$$\Delta Q_1 = Q_{1NR} - Q_{1R} > 0, \quad \Delta L = L_{NR} - L_R > 0, \\ \Delta Q_2 = Q_{2NR} - Q_{2R} > 0 \tag{1}$$

Being $L_{NR} = Q_{1NR} - Q_{2NR}$ and $L_R = Q_{1R} - Q_{2R}$, it is immediate to state that:

$$\Delta L = \Delta Q_1 - \Delta Q_2. \tag{2}$$

This last relation is significant, since it can be considered as the energy balance of a thermodynamic cycle.

Consequently, as shown by Fig. 1, from a global energy balance point of view, the R plant behaves like a hybrid plant composed of the NR plant and n heat pumps (reverse cycles – RCs) that globally use the work ΔL to "pump" the heat ΔQ_2 from the condenser to higher temperatures, in order to supply the ΔQ_1 heat to the feedwater. It is possible to easily show a graphical interpretation of this concept only for the case of a regenerative steam cycle characterized by one steam bleeding, as depicted in Fig. 2. In particular, Fig. 2a shows a regenerative Rankine cycle on the temperature – total entropy plane: the cycle is characterized by an unitary mass flowing into the condenser and one steam bleeding (having a flow rate indicated with m) extracted from the turbine in point 5 (saturated conditions). On the other hand, Fig. 2b shows the areas which define the quantities ΔL, ΔQ_1 and ΔQ_2 on the plane having as ordinate the temperature and as abscissa the product between the bled steam mass and the specific entropy. It is important to note that the sum between the green area (that is the work input ΔL of the heat pump) and the light yellow area (that is ΔQ_2) in Fig. 2b is equal to the quantity ΔQ_1 also shown by the light blue area depicted in Fig. 2a.

Therefore the adoption of n reverse cycles can be considered, as well as the thermal regeneration, a possible technique to increase the efficiency of the steam power plant. It is then interesting to express the regenerative cycle efficiency, that is equal to the hybrid plant efficiency, as a function of the two quantities ΔL and ΔQ_1:

$$\eta_R = \frac{L_R}{Q_{1R}} = \frac{L_{NR} - \Delta L}{Q_{1NR} - \Delta Q_1} = \eta_{NR} \cdot \left(\frac{1 - \Delta L / L_{NR}}{1 - \Delta Q_1 / Q_{1NR}} \right) \tag{3}$$

where η_{NR} is the non-regenerative cycle efficiency equal to the ratio between L_{NR} and Q_{1NR}.

For the n reverse cycles a "global coefficient of performance" can be defined, namely:

$$COP_{RC} = \frac{\Delta Q_1}{\Delta L} \tag{4}$$

and so, the use of the n heat pumps permits to increase the hybrid plant efficiency, with respect to the non-regenerative plant efficiency, only if the following inequality is verified:

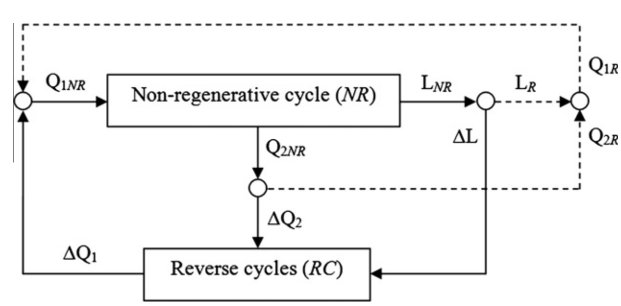

Fig. 1. A non-conventional interpretation of a regenerative steam cycle.

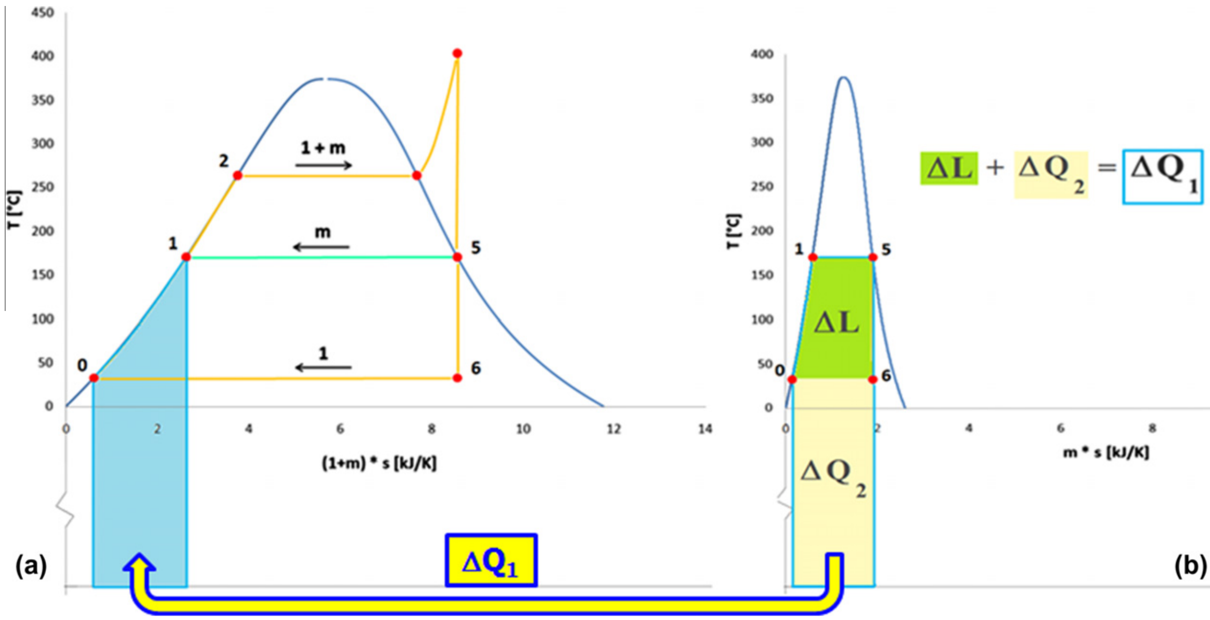

Fig. 2. A graphical interpretation of a regenerative steam cycle characterized by one steam bleeding.

$$\eta_R > \eta_{NR} \Longleftrightarrow COP_{RC} = \frac{\Delta Q_1}{\Delta L} > \frac{Q_{1NR}}{L_{NR}} = \frac{1}{\eta_{NR}} = \varphi \tag{5}$$

or, said in other words, only if ΔQ_1 is higher than the heat quantity $\Delta L/\eta_{NR}$ necessary to the non-regenerative plant to produce the work ΔL. It is also possible to write the quantity ΔQ_1 as a function of the fractional enthalpy rise R (called "regeneration degree") and the quantity λ:

$$\Delta Q_1 = \left(1 + \sum_{i=1}^{n} m_i\right) \cdot R \cdot \lambda \tag{6}$$

where m_i is the i^{th} mass of bled steam in the R cycle and, as suggested in [1,2,7,12], R expresses the enthalpy range of feedwater pre-heating non-dimensionally as a fraction of the greatest possible pre-heating range λ:

$$R = \frac{h_{a_n} - h_1}{h_2 - h_1}, \lambda = h_2 - h_1 \tag{7}$$

being h_1 and h_2 the water enthalpy values respectively at the feedwater pump exit and at the economiser outlet; moreover, h_{an} is the water enthalpy at the boiler inlet in the R plant.

Taking into account Eq. (6) and the definition of COP_{RC}, the regenerative cycle efficiency η_R (expressed by Eq. (3)) becomes:

$$\begin{aligned}\eta_R &= \frac{1}{COP_{RC}} \cdot \frac{L_{NR} \cdot COP_{RC} - (1 + \sum_{i=1}^{n} m_i) \cdot R \cdot \lambda}{Q_{1NR} - (1 + \sum_{i=1}^{n} m_i) \cdot R \cdot \lambda} \\ &= \frac{\eta_{NR}}{COP_{RC}} \cdot \frac{COP_{RC} - (1 + \sum_{i=1}^{n} m_i) \cdot R \cdot \lambda \cdot L_{NR}^{-1}}{1 - (1 + \sum_{i=1}^{n} m_i) \cdot R \cdot \lambda \cdot Q_{1NR}^{-1}}.\end{aligned} \tag{8}$$

Thus, the plant efficiency dimensionless gain, due to feedwater heating, can be calculated as:

$$\frac{\Delta\eta}{\eta} = \frac{\eta_R - \eta_{NR}}{\eta_{NR}} = \frac{\Delta Q_1}{Q_{1R}} \cdot \left(1 - \frac{1}{\eta_{NR} \cdot COP_{RC}}\right). \tag{9}$$

It derives that, a certain water mass flow entering the boiler being given, the regenerative cycle efficiency gain is only a function of the coefficient COP_{RC} and of the ratio R, this last influencing ΔQ_1 by means of Eq. (6). Moreover, for a given value of R, the plant efficiency maximum increase is attained when the COP_{RC} factor is maximized.

3. The regenerative steam power plant

To reinforce and apply the considerations and formulas previously exposed (Section 2), these last have been applied to a regenerative Rankine steam cycle with two feedwater heaters; the scheme of the power plant is reported in Fig. 3.

Some assumptions have been made in order to precisely control the energy balances. In most calculations regarding steam cycles regeneration, the work exchanged with fluid in the liquid phase is often neglected in energy balances, being very small with respect to that exchanged in the vapor phase. In the present calculations, instead, the work related to the liquid phase, exchanged respectively in the hydraulic turbines and the feedwater pump, has been considered, leading to the demonstration of the energy balance closure also including the incompressible fluid work.

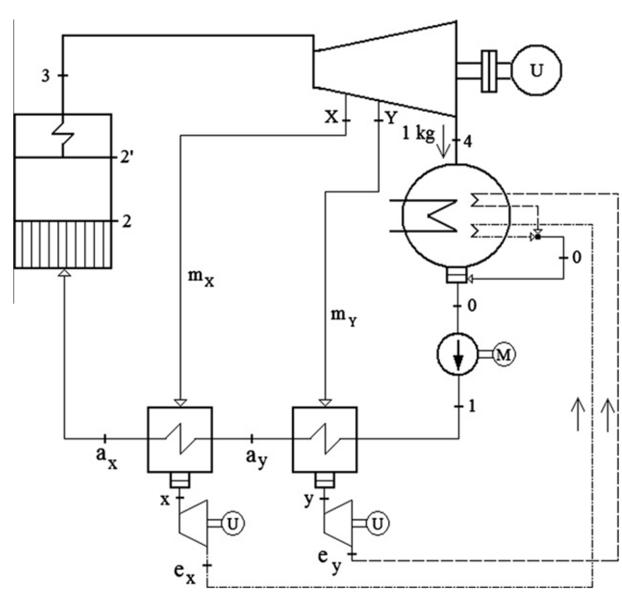

Fig. 3. Conceptual scheme of the regenerative steam power plant.

Going into details, the main hypotheses assumed for the present study are:

- Heat losses and pressure drops in the boiler are neglected.
- All the machinery composing the power plant is considered ideal (unitary isentropic efficiency) except the steam turbine. In fact the steam turbine efficiency η_T has been varied to investigate its relative influence on the monitored variables.
- The feedwater heaters are assumed to have a null pinch point temperature difference between the two fluids (infinite heat exchange area).
- The steam bleedings are effected at expansion pressures sufficiently low to have the point X and Y below the upper limit curve, that is into the wet steam zone in the Mollier chart, so that the bled streams can exchange heat with the feedwater maintaining a constant temperature.
- The feedwater heaters are dimensioned to cause bleedings complete condensation without subcooling.

As visible in Fig. 3, a water mass of $(1 + m_Y + m_X)$ kilograms at state a_x enters the boiler, exits from the economiser as saturated liquid (state 2), evaporates and finally is superheated; the steam supplied by the boiler (state 3) expands through the turbine; the steam masses m_X and m_Y are sequentially extracted (X and Y points) from the expanding flow to provide the regeneration heat. The remaining unitary mass of steam, not bled, enters the condenser (state 4) being transformed into saturated liquid at state 0. The bled steam quantities m_X and m_Y condense in two surface heat exchangers, providing heat to the pumped feedwater, which turns from state 1 to state a_y in the first heater and then to state a_x in the second one; the water drained from the two feedwater heaters, respectively at the states y and x, transforms its pressure energy into work crossing two hydraulic turbines which expand, isentropically, the saturated liquid (at the two bleeding pressures) to the condenser pressure (state e_y and e_x); this expansion causes the formation of saturated vapor in the streams (quality higher than 0 and lower than 1), therefore the two quantities m_Y and m_X are conveyed to the condenser where they reach the saturation state, named 0 (the same state as the main-stream water leaving the condenser), exchanging heat with the cooling water. The three condensed streams are then mixed in the condenser hot-well and pumped to the feed-water pressure (state 1).

The states assumed by the fluid along the regenerative Rankine cycle are represented in Fig. 4 on the Mollier chart, and a zoomed view evidencing the transformations in the liquid phase is provided in Fig. 5.

In an actual plant, the drain water coming from the ith turbine bleeding, collected in the ith feedwater heater hot-well is often expanded through a throttling valve and then mixed with the feedwater. In the examined conceptual plant, in order to have complete reversibility as suggested by Haywood in [1], the condensed bled steam from each heater is cascaded down to the condenser via a reversible drain-water turbine, otherwise there would be a lost opportunity for producing work and the energy balance on which the present approach is based would be incomplete.

The bled steam masses m_Y and m_X have been calculated by the energy balance applied to the two feedwater heaters, reported in Fig. 3, and they are given by:

$$m_X = \left[\frac{\frac{H_X - h_x}{h_{a_x} - h_{a_y}}}{1 + \frac{1}{\frac{H_Y - h_y}{h_{a_y} - h_1} - 1}} - 1 \right]^{-1}, \quad m_Y = \frac{1 + m_X}{\frac{H_Y - h_y}{h_{a_y} - h_1} - 1} \tag{10}$$

4. The hybrid steam power plant

As stated in the previous paragraphs, thermal regeneration can be conceived as provided by reverse cycles, which "pump" the heat ΔQ_2 from the condenser to the feedwater, as ΔQ_1, exploiting the work ΔL gained by the expansion in turbine of m_X and m_Y mass flows. Fig. 6 represents the conceptual plant, called "hybrid steam power plant", related to the above mentioned solution.

The thermodynamic system examined can be considered as a hybrid system constituted by:

- a Hybrid Engine Cycle (HEC): it is the non-regenerative Rankine cycle 0–1-a_y–a_x–2–2′-3–4, in which $(1 + m_Y + m_X)$ kilograms of water circulate;
- a Reverse Cycle (RC) Y: it is the 4-Y–y–e_y cycle to which the water mass m_Y is subjected;
- a Reverse Cycle (RC) X: it is the 4-X-x-e_x cycle to which the water mass m_X is subjected.

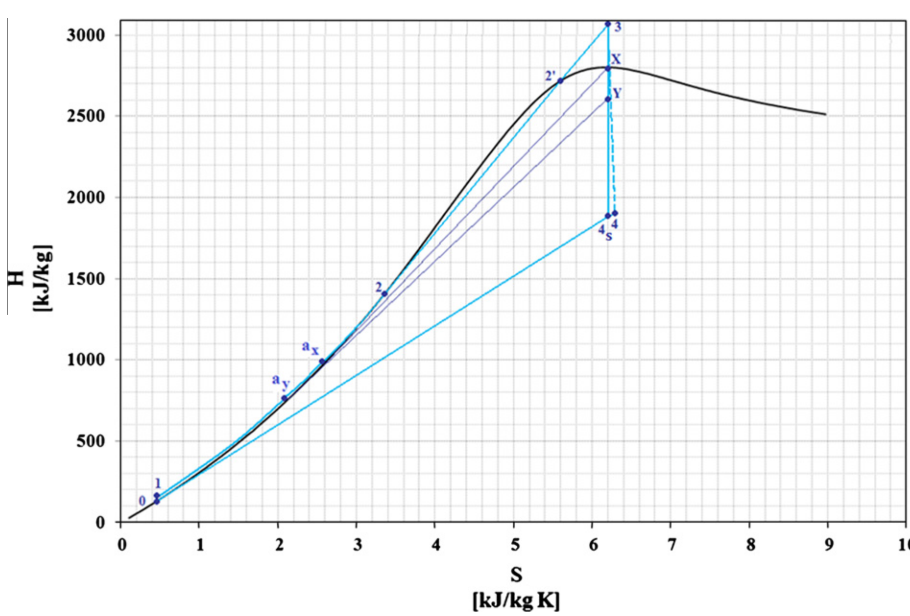

Fig. 4. Regenerative Rankine cycle on the enthalpy–entropy plane.

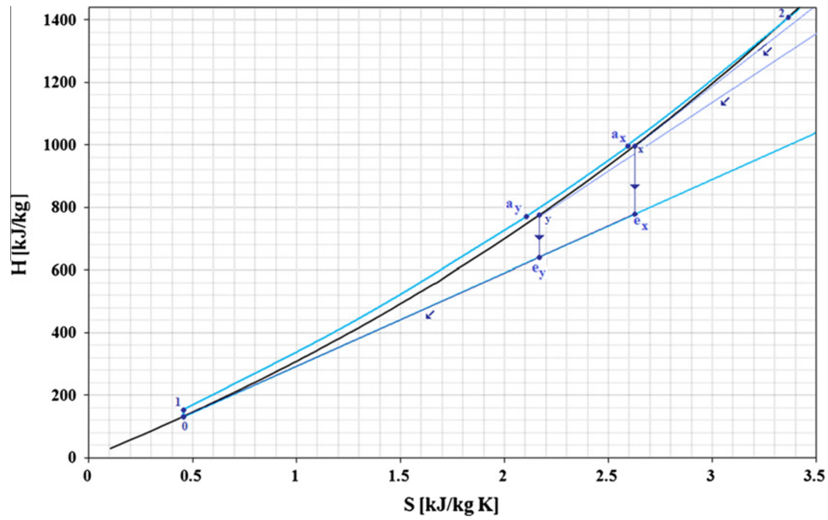

Fig. 5. Zoomed view of the regenerative Rankine cycle at low enthalpies.

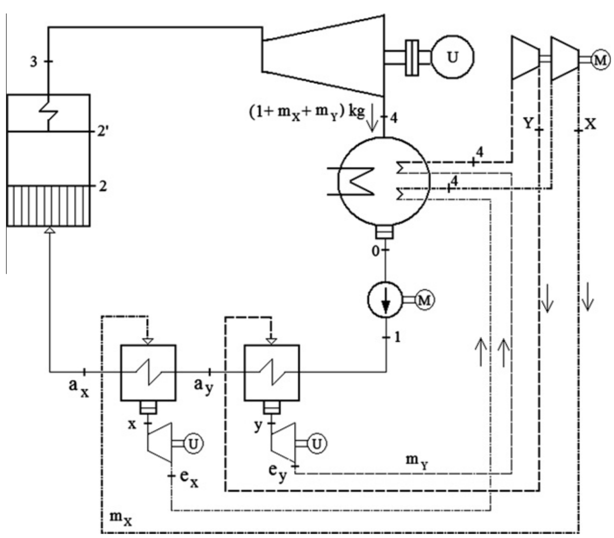

Fig. 6. The hybrid steam power plant.

In these last two reverse cycles, water absorbs heat (respectively Q_{2Y} and Q_{2X}, being $\Delta Q_2 = Q_{2Y} + Q_{2X}$) from the condenser reaching state 4. Fluid is then compressed respectively to state Y (lower pressure) and state X (higher pressure). The compressed fluid, that is wet vapor, is delivered to the surface feedwater heaters, where condenses providing heat (respectively Q_{1Y} and Q_{1X}, being $\Delta Q_1 = Q_{1Y} + Q_{1X}$) to the feed-water whose enthalpy globally increases from h_1 to h_{ax}. Then the two mass quantities m_Y and m_X expand through two hydraulic turbines from which exit at e_y and e_x thermodynamic conditions. The thermodynamic transformations for the Hybrid Engine Cycle and the two Reverse Cycles may be visualized in Fig. 4 and 5, being the thermodynamic states of the working fluid the same as those indicated for the regenerative cycle.

4.1. The global COP calculation

Considering a hybrid power plant such as that reported by Fig. 6, but characterized by n heat pumps, it is interesting to observe that the quantities ΔQ_1, ΔQ_2 and ΔL can be calculated as:

$$\Delta Q_1 = \sum_{i=1}^{n} Q_{1i},\ \Delta Q_2 = \sum_{i=1}^{n} Q_{2i},\ \Delta L = \sum_{i=1}^{n} L_i \tag{11}$$

where Q_{1i}, Q_{2i} and L_i are the energy quantities relative to the ith heat pump, which is characterized by its "coefficient of performance" equal to:

$$COP_{RCi} = \frac{Q_{1i}}{L_i} \tag{12}$$

It is possible to express the global "coefficient of performance" COP_{RC} of the hybrid cycle as a function of the n COP_{RCi} values obtaining, after a little algebra, the following interesting relation:

$$COP_{RC} = \frac{\Delta Q_1}{\Delta L} = \frac{1}{\Delta L} \cdot \sum_{i=1}^{n} (COP_{RCi} \cdot L_i) \tag{13}$$

which shows that COP_{RC} is the average of the n COP_{RCi}, each weighted with its corresponding $L_i/\Delta L$ ratio.

Finally, it is remarkable to point out that for a regenerative steam cycle with n bleedings characterized by unitary steam turbine isentropic efficiency and non-superheated bled steam, the n reverse cycles are Carnot reverse cycles.

5. Results of the calculations

The "base" thermodynamic cycle chosen for the calculations is a Rankine cycle having evaporation pressure of 100 bar, maximum temperature of 400 °C and condensing pressure of 0.05 bar.

The systems described in the previous paragraph, namely the regenerative plant and the hybrid plant respectively shown in Figs. 3 and 6, have been compared from the energetic point of view to demonstrate their substantial thermodynamic equivalence.

5.1. Energy balances

In Fig. 7 the bar charts representing the energy balance of the thermodynamic cycles investigated are depicted. The results refer to optimal bleedings pressures, calculated according to the simplified Salisbury's theory, and unitary steam turbine efficiency.

The diagrams reported in Fig. 7 represent, concisely, the energy balance of the three analyzed power plants:

- "Base Cycle", constituted by the non-regenerative Rankine cycle.
- "Regenerative Cycle", represented by the plant shown in Fig. 3.
- "Hybrid Cycle" (CHC), represented by the plant shown in Fig. 6.

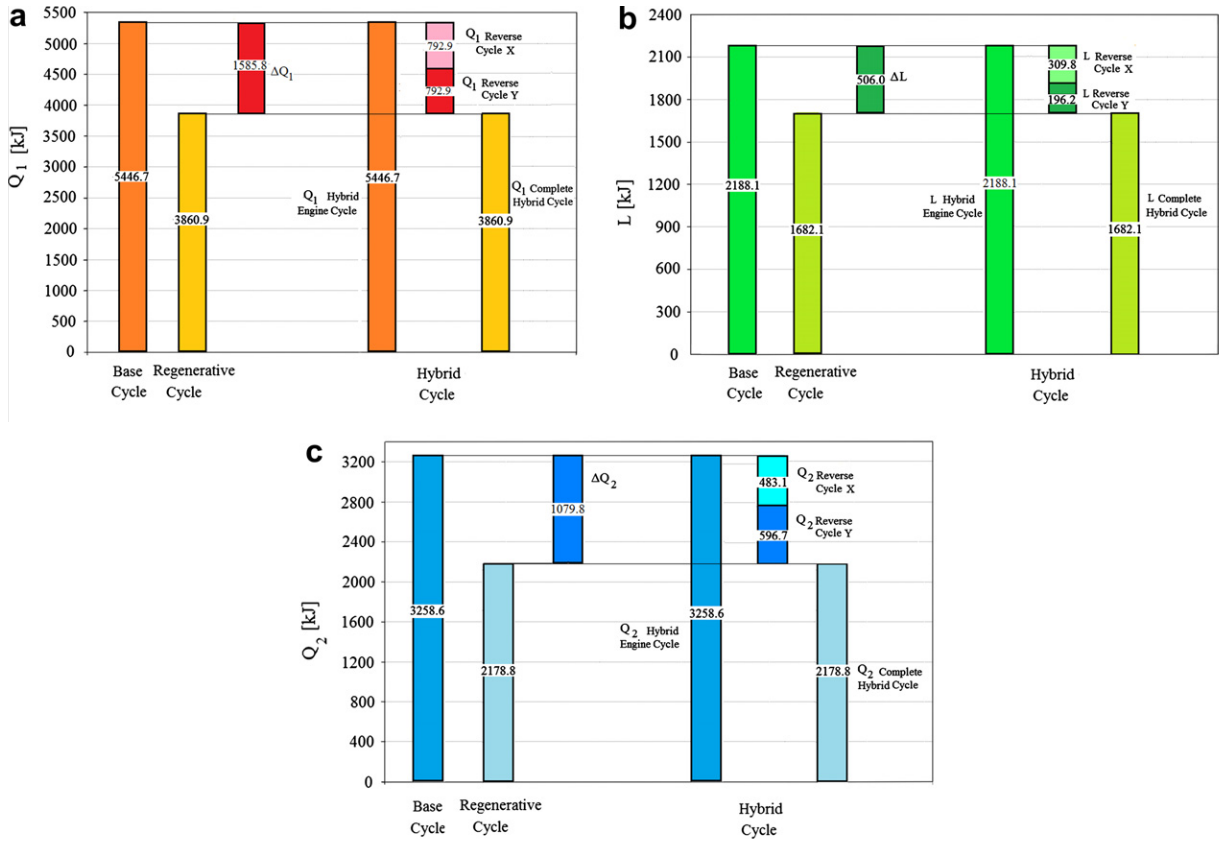

Fig. 7. Energy balances for the compared thermodynamic cycles: (a) Q_1; (b) L; (c) Q_2.

The Q_1 diagram is presented in Fig. 7a. In accordance to Eq. (11), the heat difference ΔQ_1 between the Q_1 of the Base Cycle and the Q_1 of the Regenerative Cycle is equal to the sum of the heat quantities (indicated in Fig. 7a as "Q_1 Reverse Cycle *Y*" and "Q_1 Reverse Cycle *X*") provided to the feedwater by the two reverse cycles. Furthermore, the Q_1 heat supplied to the regenerative cycle (equal to 3860.9 kJ) is the same as that provided to the CHC hybrid cycle, both cycles having the same boiler inlet and outlet conditions (points a_x and 3).

As previously stated in Section 4, the CHC hybrid cycle can be considered as the composition of three thermodynamic cycles; from this point of view, the Q_1 amount supplied to the "Hybrid Engine Cycle" (equal to 5446.7 kJ) can be considered as the heat provided in the boiler (indicated in the figure as "Q_1 Complete Hybrid Cycle") plus the heat provided by the two heat pump cycles ("Q_1 Reverse Cycle *Y*" and "Q_1 Reverse Cycle *X*"). It is important to point out that the Q_1 amounts of the two reverse cycles are equal (792.9 kJ), as stated by the Salisbury's theory according to which the enthalpy rise has to be the same in all feedwater heaters [1,7].

The calculated work *L* for each cycle is shown in Fig. 7b. As proved for ΔQ_1, the work difference ΔL between the *L* of the Base Cycle and the *L* of the Regenerative Cycle, is equal to the sum of the work inputs of the "Reverse Cycle *Y*" and the "Reverse Cycle *X*".

The net work of the Complete Hybrid Cycle (equal to 1682.1 kJ) can be calculated by subtracting the work inputs of the two heat-pumps ("*L* Reverse Cycle *Y*" equal to 196.2 kJ and "*L* Reverse Cycle *X*" equal to 309.8 kJ) from the work output of the Hybrid Engine Cycle (equal to 2188.1 kJ). The "Reverse Cycle *X*" has a work input higher than that absorbed by the "Reverse Cycle *Y*" since it is characterized by a higher pressure at the compressor discharge and so by a lower *COP*.

The rejected heat Q_2 has also been analyzed, and the results are compatible with those of Q_1 and *L*. A part of the heat rejected by the "Hybrid Engine Cycle" constitutes the heat input of the two reverse heat pump cycles, equal to ΔQ_2.

5.2. Parametric analysis

Several tests have been carried out to determine the variation of the regenerative plant main parameters as a function of the bleedings pressure values. In particular, the tests have been conducted maintaining constant the pressure of the first bleeding (p_X), and varying the pressure of the second bleeding (p_Y). The following diagrams, representing the monitored variables, have in abscissa the partial degree of regeneration $R_Y = (h_{ay} - h_1)/\lambda$.

Fig. 8 shows the steam masses (m_Y and m_X) as a function of R_Y. Different curves have been sketched varying the turbine isentropic efficiency η_T from 0.86 to 1. As visible, the two quantities m_Y and m_X are almost the same when R_Y is equal to ca. 0.33 (Salisbury's optimal condition) and, for a given R_Y, they increase with η_T.

Figs. 9–11 represent the regenerative power plant energy flows Q_1, *L* and Q_2 normalized with respect to their values Q_{1opt}, L_{opt} and Q_{2opt}, calculated considering the Salisbury optimal values for p_Y and p_X, diagrammed as a function of R_Y. Also in this case, different turbine efficiencies η_T in the range [0.86;1] have been considered.

As visible in Figs. 9–11, Q_1/Q_{1_opt} and L/L_{opt} curves show a certain spread, for the different isentropic turbine efficiencies, far from the condition $R_Y = 0.33$, whereas Q_2/Q_{2_opt} curves display a smaller variation with η_T.

Fig. 12 reports the regenerative cycle efficiency, equal to that of the complete hybrid cycle, as a function of R_Y for the various turbine efficiency values. Obviously, lower turbine efficiencies correspond to lower cycle efficiencies and, for all the curves, the maximum efficiency is set at about R_Y equal to 0.29, lower than the optimal Salisbury value (0.33). This is mainly imputable to the fact that the simplified analytical Salisbury's theory is formu-

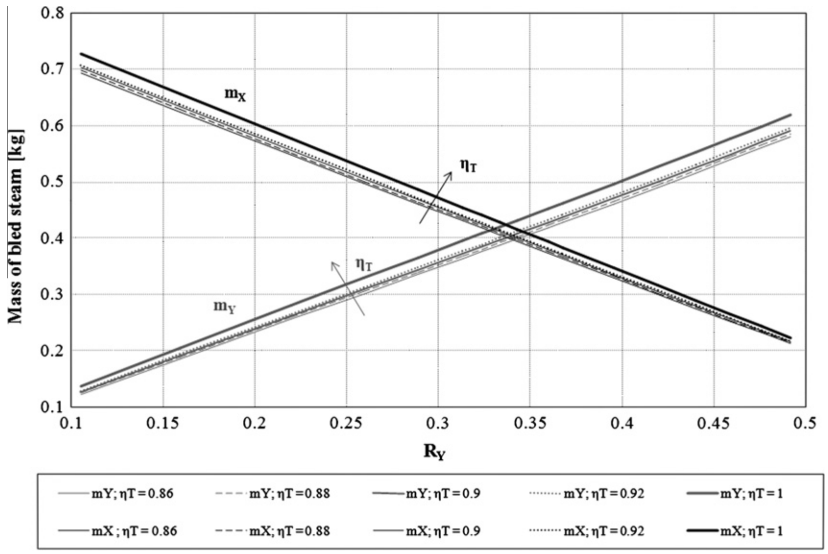

Fig. 8. Bled steam quantities as a function of R_Y.

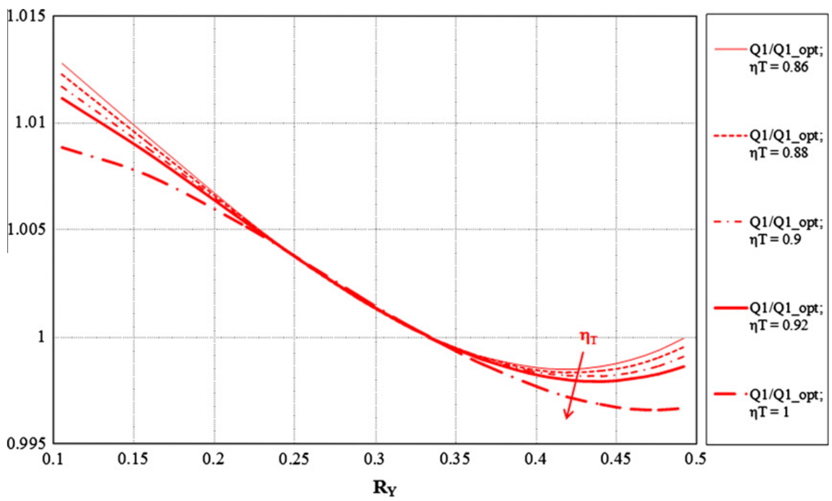

Fig. 9. The Q_1/Q_{1opt} ratio as a function of R_Y.

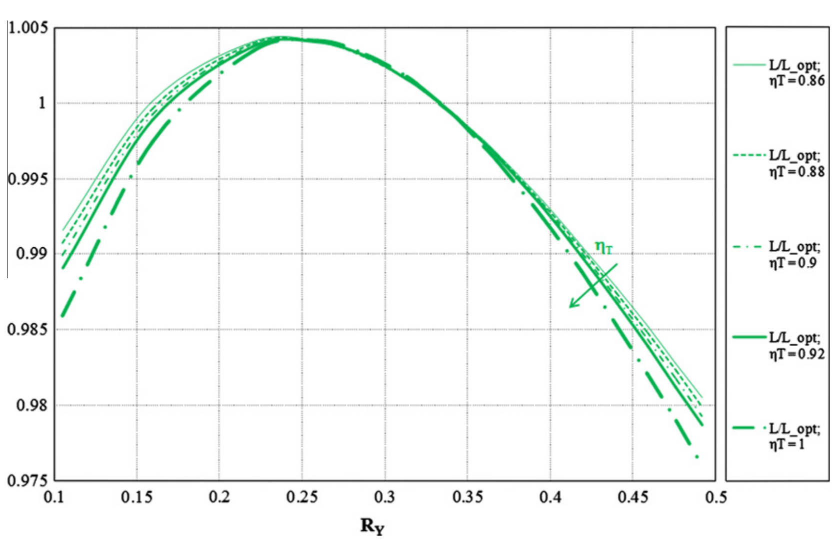

Fig. 10. The L/L_{opt} ratio as a function of R_Y.

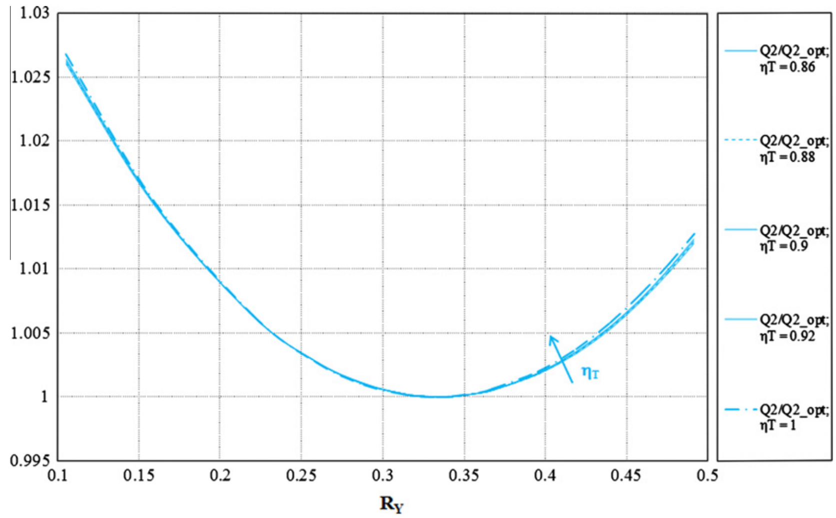

Fig. 11. The Q_2/Q_{2opt} ratio as a function of R_Y.

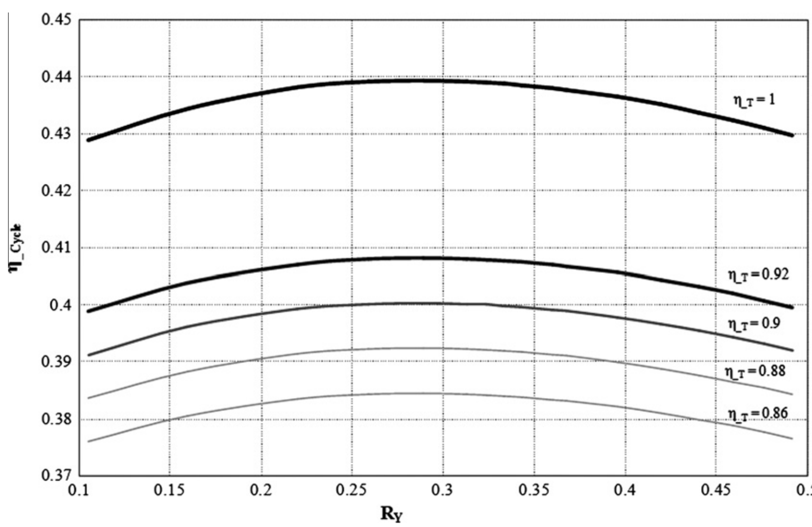

Fig. 12. The regenerative and CHC cycles efficiencies as a function of R_Y.

lated in the hypothesis of considering contact feedwater heaters and, at any bleeding pressure, a constant enthalpy difference between bled steam and saturated liquid; moreover, the work related to machinery elaborating incompressible fluid is not taken into account. Evidently, overtaking these simplifications leads to different results.

In Fig. 13 the *COP* values for the two Reverse Cycles *X* and *Y* constituting the hybrid plant are graphed as a function of R_Y and η_T. By hypothesis, the Reverse Cycle *X* operates at a constant pressure, therefore its *COP* value remains constant. Instead, the *COP* of the Reverse Cycle *Y* changes with R_Y, and so with p_Y, assuming very high values for p_Y approaching the condenser pressure; as p_Y tends to p_X, COP_Y tends to COP_X. These observations are in accordance with reverse cycles theories, for which a reverse cycle operating between two very close temperature values is characterized by a very high *COP*.

The global *COP*, calculated as a function of COP_X and COP_Y by Eq. (13), is shown in Fig. 14. The global *COP* has maximum values, for all the expansion efficiency curves, at R_Y almost equal to 0.29, as found for the maximum efficiency of the regenerative Rankine cycle. Fig. 14 also displays the φ values that, in accordance with Eq. (5), are the reciprocal values of the non-regenerative cycle efficiency; as visible in Fig. 14, the inequality reported in Eq. (5) is always satisfied.

5.3. A possible application

A possible application of the presented theory to the industrial reality could be feasible for the small electrical power production plants. These ones are seldom regenerated, due to the increase in complexity and costs that thermal regeneration would imply. Nevertheless, in some cases, a pressurized fluid reservoir at equilibrium conditions is needed, above all for degassing purposes. To avoid the solution of a steam bleeding from the turbine, which would require the customization (and therefore the increase in costs) of the turbomachinery, one solution could be a live steam bleeding, as represented in Fig. 15a (on the left). Obviously, this solution would not increase the plant efficiency.

The data reported in Fig. 15 are referred to a calculation made in the hypothesis of utilizing a Fincantieri MS–VS turbine, without steam bleedings, for power production.

As an alternative to the live steam bleeding, a solution consisting in a hybrid steam plant with a water heat pump reverse cycle, pumping heat from the condenser to the hot fluid reservoir, might

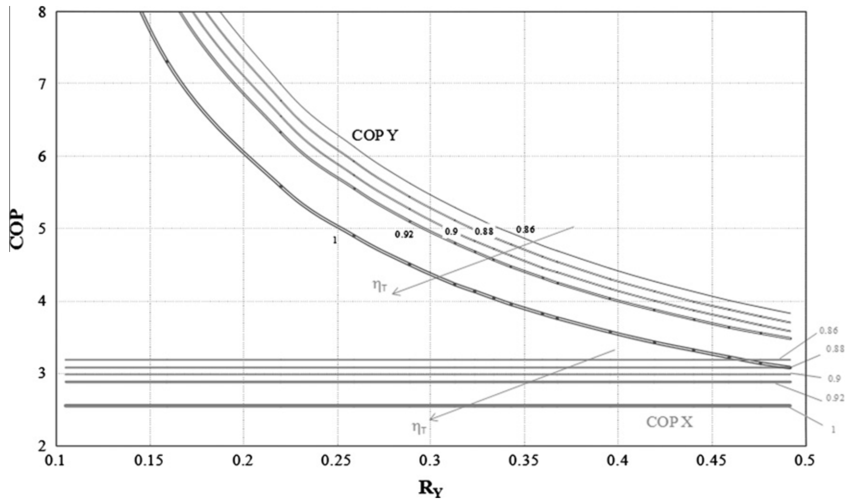

Fig. 13. The reverse cycles COP values as a function of R_Y.

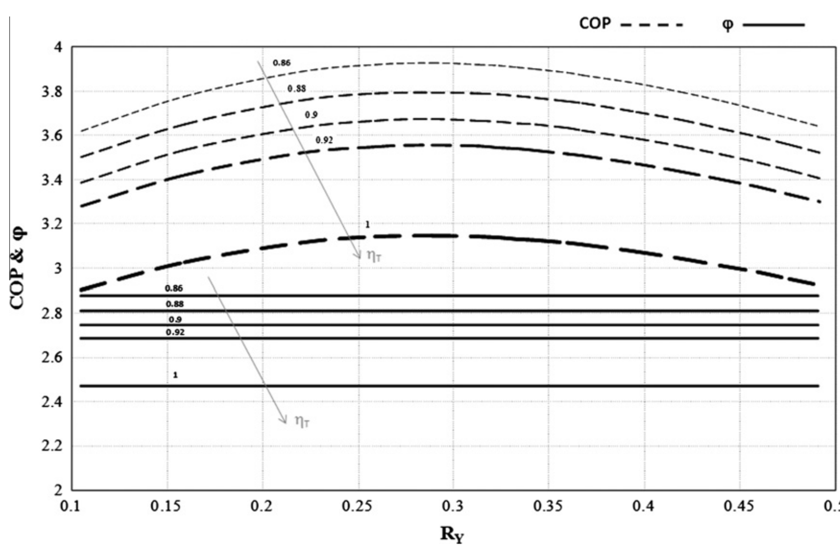

Fig. 14. Global COP and as a function of R_Y.

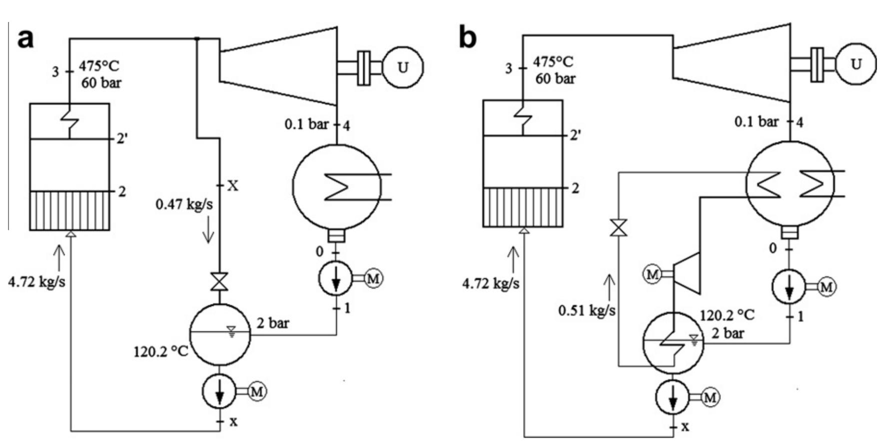

Fig. 15. A possible application of the presented theoretical method to an existing small power plant: (a) conventional solution and (b) solution with heat pump from the condenser.

be considered, as shown by Fig. 15b. In this case, an increase in the plant efficiency is expected, as shown in Table 1, in which the overall performances of the conventional and of the hybrid plants are presented.

Table 1
Performances of the live steam bleeding and hybrid power plants.

	Conventional solution (Fig. 15a)	Solution with heat pump (Fig. 15b)
Turbine power	4577.49 kW	5078.18 kW
Pumps power	35.07 kW	35.16 kW
Heat pump compressor power	–	421.71 kW
Thermal power input	13463.78 kW	13463.78 kW
Efficiency	0.337	0.343

The hybrid plant efficiency increase is not certainly large enough to justify the increase in capital costs (mainly due to the steam compressor, characterized by a significant volumetric flow rate) determined by the heat pump installation. However, the present study is not aimed at estimating the feasibility of a possible installation from the economic point of view; the investigation of different working fluids for the heat pump might be an interesting development of the present study, in order to find a technical solution able to reduce the volumetric flow rate in the heat pumps and, consequently, the compressor capital cost.

6. Conclusion

In the presented paper, the authors have proposed a non-conventional approach to face the technique of thermal regeneration, a powerful mean to increase the efficiency of steam power plants. This approach allows to analyze, from the thermodynamic point of view, a regenerative Rankine cycle as a hybrid power plant constituted by one Rankine cycle without steam bleedings and n heat pumps that contribute to the feedwater enthalpy rise.

To validate the developed theoretical relations, the analysis of a regenerative steam cycle has been carried out, applying the proposed theory to a Rankine cycle with two steam bleedings. The application includes the study of the plant energy balances taking into account different values of the turbine isentropic efficiency and not neglecting the work of incompressible fluid machinery. The presented approach has allowed the authors to draw several interesting results and considerations on the *COP* of the global heat pump system and to propose some hints for a possible application.

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